REMARKS/ARGUMENTS

STATUS OF CLAIMS

Claims 1-20 are currently pending in this application. By this Amendment, claims 1, 15, and 17 are amended, leaving claims 2-14, 16, and 18-20 unchanged.

CLAIM REJECTIONS – 35 U.S.C. § 102

On page 2 of the Office Action, claims 1 and 17 are rejected under 35 U.S.C. §102(a) as being anticipated by Khrustalev et al. (U.S. Patent No. 6,536,510).

Claim 1 is hereby amended, and calls for:

> A capillary assisted loop thermosiphon apparatus comprising:
> at least one evaporator connected by a vapor line to a condenser;
> a liquid line connecting the condenser and the evaporator;
> the evaporator is vertically elongated and is positioned in the direction of gravity from the condenser such that the condenser supplies liquid under gravity induced pressure to the evaporator, and the evaporator has a vertical capillary wick in which liquid wicks in the direction of gravity. (Amendment marks not shown).

Claim 17 is also hereby amended, and calls for:

> A capillary assisted loop thermosiphon apparatus comprising:
> at least one evaporator connected by a vapor line to a condenser;
> a liquid line connecting the condenser and the evaporator;
> the evaporator is positioned in the direction of gravity from the condenser such that the condenser supplies liquid under gravity induced pressure to the evaporator; and
> the evaporator has at least a pair of vertically elongated sheets, with at least one of the sheets having a corresponding wick portion attached thereto to provide a vertical capillary wick in which liquid wicks in the direction of gravity."
> (Amendment marks not shown).

In contrast, Khrustalev et al. disclose a thermal bus that includes "two spaced-apart substantially horizontally oriented evaporators ... provided where each is substantially horizontally mounted in a support and positioned in thermal communication with at least one heat generating device. Each of the two elongate evaporators defines a central passageway having a liquid-working fluid entrance port and a vaporous-working fluid exit port and a capillary wick disposed on the walls of the central passageway." (Col. 3, lines 20-30)

Indeed, Khrustalev et al. actually <u>teach away</u> from a capillary assisted loop thermosiphon apparatus having a vertically elongated evaporator or having an evaporator with a pair of vertically elongated sheets as claimed in amended claims 1 and 17, because Khrustalev et al. make clear that in the disclosed horizontal orientation of the Khrustalev et al. device, the working fluid flows along each horizontally arranged rail evaporator mainly due to the frictional vapor-liquid interaction on the liquid free surface of the central passageway... ensuring a very low thermal resistance of rail-evaporators, in comparison to vertically oriented thermosyphon evaporators. (Col. 6, lines 59-67). Clearly, Khrustalev et al. teach a horizontally-extending evaporator for a very specific purpose, and teach away from an evaporator as claimed in amended claims 1 and 17.

Accordingly, Khrustalev et al. fail to teach, describe, or suggest, among other things, a capillary assisted loop thermosiphon apparatus comprising at least one vertically elongated evaporator positioned in the direction of gravity from a condenser, and having a vertical capillary wick as claimed in amended claim 1. Similarly, Khrustalev et al. fail to teach, describe, or suggest, among other things, a capillary assisted loop thermosiphon apparatus comprising at least one evaporator positioned in the direction of gravity from a condenser and having at least a pair of vertically elongated sheets, either or both of which are provided with a vertical capillary wick as claimed in amended claim 17.

In light of these and other reasons not discussed herein, the Applicants respectfully submit that independent claims 1 and 17 are novel and non-obvious over Khrustalev et al. Withdrawal of the 35 U.S.C. §102(a) rejections of claims 1 and 17 in view of Khrustalev et al. is therefore respectfully requested.

Also on page 2 of the Office Action, claims 2, 3, 5, 11, 12, 18, and 19 are rejected under 35 U.S.C. §102(a) as being anticipated by Khrustalev et al. Claims 2, 3, 5, 11, and 12 depend from claim 1 and are therefore allowable based upon amended claim 1, and for other reasons not discussed herein. Claims 18 and 19 depend from claim 17, and are therefore allowable based upon amended claim 17 and for other reasons not discussed herein. Withdrawal of the 35 U.S.C. §102(a) rejections of claims 2, 3, 5, 11, 12, 18, and 19 is therefore respectfully requested.

On pages 2 and 3 of the Office Action, claims 1-20 are rejected under 35 U.S.C. §102(a) as being anticipated by Conroy et al. (Multiple Flat Plate Evaporator Loop Heat Pipe Demonstration). Enclosed with this Amendment is a copy of the reference, which is clearly dated August 17-21, 2003. The present application was filed March 19, 2004, and claims priority to U.S. provisional patent app. no. 60/456,262 filed on March 20, 203. Accordingly, the Applicants respectfully submit that the Conroy et al. referenced cited on page 3 of the Office Action is not prior art to the claims of the present application. Accordingly, withdrawal of the 35 U.S.C. §102(a) rejection of claims 1-20 in view of Conroy et al. is respectfully requested.

Claims 2-16 and 18-20 are each ultimately dependent upon amended claims 1 and 17, respectively, and are allowable based upon amended claims 1 and 17 and upon other features and elements claimed in amended claims 1 and 17 but not discussed herein. Withdrawal of the 35 U.S.C. §102(a) rejections of claims 2-16 and 18-20 is therefore respectfully requested.

CONCLUSION

In view of the above, Applicants respectfully request entry of the Amendment and allowance of pending Claims 1-20.

Respectfully submitted,

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MULTIPLE FLAT PLATE EVAPORATOR LOOP HEAT PIPE DEMONSTRATION

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ABSTRACT

A gravity assist loop heat pipe system suitable for cooling large arrays of electronic devices has been designed, fabricated, and successfully tested. An envisioned application dictated the geometry of the evaporators as flat plates, 24.4 cm by 76.2 cm, and required a total of 12 kW of heat per plate to be removed. One important aspect of the experiment was whether it is practical to network several evaporators together to utilize a common, water-cooled condenser. A combination of four evaporators was chosen to simulate a building block for a larger system, providing a total of 48 kW of heat removal. This paper describes the design and test results to-date of that unit.

OVERVIEW OF CAPILLARY PUMPED TECHNOLOGY

The basic operation of a loop heat pipe (LHP) is shown schematically in Figure 1. Basically, the system provides a passive means of transporting heat from the evaporator section to a remote condenser. The devices do not have any mechanical moving parts to wear out, require no electrical power to operate, and demand virtually no maintenance. Heat is absorbed in the evaporator section, vaporizing the system working fluid. A porous material, or wick, in the evaporator provides the pumping power from surface tension forces developed in the surface of the wick. The vapor transports to the remote condenser, where it is condensed and typically slightly subcooled. Liquid then returns to the evaporator to complete the cycle.

In some applications as shown in Figure 1, a reservoir physically attached to the evaporator serves as a means of fluid management. The liquid return line passes through the reservoir that is physically attached to the evaporator. This arrangement ensures that the reservoir is cold-biased. Thus, fluid will collect in this volume regardless of gravity conditions. In addition, excess vapor from the evaporator collects in the reservoir through a secondary wick. The liquid line that passes through the reservoir contains subcooled liquid which condenses the excess vapor. This, in turn, ensures liquid is available to the evaporator during power transients. The disadvantage of this arrangement is a complicated manufacturing process and system inefficiency in that heat is lost to the reservoir rather than transferred to the condenser.

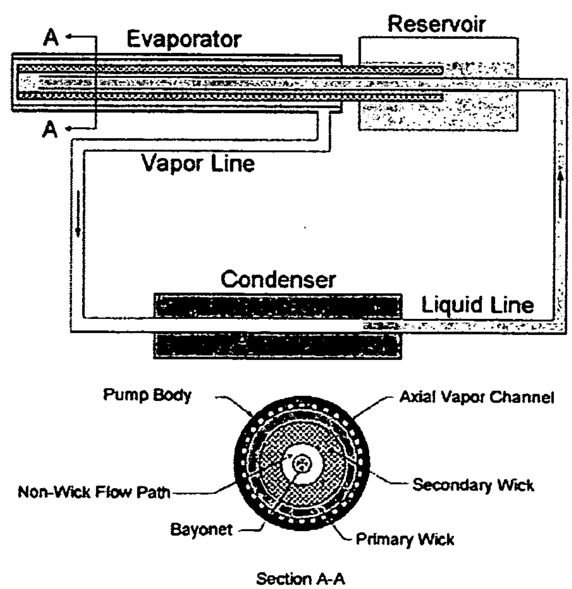

Figure 1 – Schematic of Typical Loop Heat Pump

MULTIPLE FLAT PLATE EVAPORATOR LOOP HEAT PIPE DEMONSTRATION

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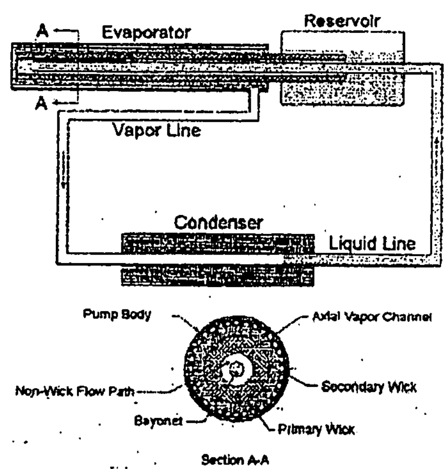

Figure 1 – Schematic of Typical Loop Heat Pump

Detailed descriptions of capillary pumping technologies, including a comparison of the advantages of LHPs and capillary pump loops are available in the literature[1, 2, and 3].

SYSTEM DESIGN CONSTRAINTS

Passive heat transport devices include heat pipes, LHPs, capillary pump loops (CPL), and loop thermosyphons. LHPs and CPLs were originally intended to provide high heat transport capability in microgravity applications. But the ability of these devices to transport high heat loads has drawn the interest of ground-based systems. In this particular study, the feasibility of using a multiple evaporator LHP system to move a large amount of waste heat to a remotely located heat exchanger was considered.

The intended application for this study required mounting a large number of electrical components directly to the heat sink. The assembly would then be installed in a rack type cabinet with the condenser located above the cabinet. As a result of this arrangement, all liquid feed and vapor connections were required to enter from one side of the evaporator. This also facilitated removal of a portion of the overall system for maintenance and/or replacement. Uniformity of temperature across the entire surface of the heat sink was necessary to provide peak efficiency of the electrical components. The system is intended to operate in a gravity application, but must be capable of functioning while inclined adversely up to ±45° along the long axis as depicted in Figure 2.

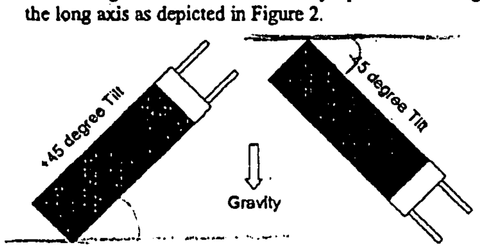

Figure 2 – Tilt Conditions of Evaporator

Consideration of a capillary pumped system was initially due to the interest in eliminating or minimizing the use of mechanical devices. Further investigation showed several other advantages to the LHP concept. First, because the system relies on the heat of vaporization of the working fluid, the heat sink surface is essentially isothermal.

Additionally, the system could be designed such that the total amount of working fluid was limited. The advantage with this is that if a portion of the system ruptures, the volume of liquid released can be more readily accommodated than with a conventional cooling water system.

A compromise between electrical system arrangement, minimizing the overall volume of the completed system, and the maintenance/replacement constraints identified that the heat sink would need to be a flat plate, 24.4 cm by 76.2 cm. The thickness of the heat sink plate was also limited to 1.3 cm. The total heat load was determined to be 12 kW per plate assuming electrical devices were mounted on either side of the plate, resulting in a heat flux of 3.2 kW/cm². An active heating area was specified to cover at least 90 percent of the evaporator width as shown in Figure 3. The required transport distance is 10 feet.

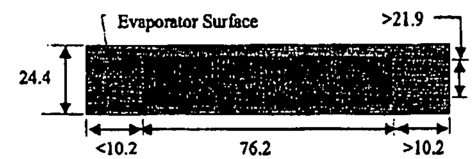

Figure 3 - Flat Rectangular Evaporator (all units in cm)

In summary, the constraints provided for the cooling system are described in Table 1.

Table 1: Design Requirements

Flat plate geometry 1.3 cm by 24.4 cm by 76.2 cm
Feed and liquid connection on one side of evaporator
Heat flux removal of 3.2 kW/cm²
Isothermal surface temperature within 3°C
Active heat removal area, 90 percent of evaporator
Heat transport distance 305 cm
Operation in gravity
Adverse inclination up to ±45°
Minimal fluid volume
Network of several evaporators to a common condenser
Maximum Operating Temperature, <65°C
Cooling Water Temperature (Sink), 2°C–32°C
Flatness Across Evaporator Width, ±0.013 cm
Straightness Along Evaporator Length, ±0.05 cm

DESCRIPTION OF DESIGN

The gravity-assist LHP concept is shown schematically in Figure 4. Recognizing that the system would only be used in terrestrial applications, gravity was relied upon for fluid management. This was accomplished by locating the condenser above the evaporator. The advantage of this arrangement is that gravity will

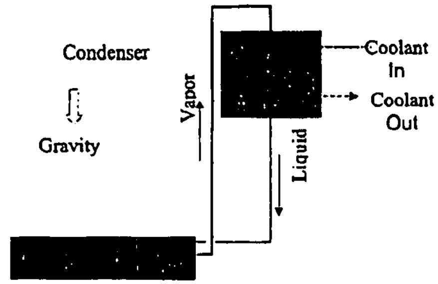

Figure 4 – Schematic of Gravity Assist Loop Heat Pipe

always ensure that the liquid returns to saturate the wick. In addition, excess fluid accumulates in the bottom of the evaporator during low power operation. With this stipulation, the complicated reservoir and secondary wick structure were not necessary for fluid management and were eliminated. Additionally, the gravitational head, created by placing the loop condenser above the evaporators, augmented the system pumping capability by almost 100 percent.

There were several areas of concern identified during the design process of this system. First of all, liquid would completely flood the evaporator internal volume (including vapor channels) before system start-up. As a result, a temperature superheat would likely occur on the evaporator surface for nucleate boiling to initiate. Boiling with a high temperature superheat would induce a stressful flow condition that might lead to a wick deprime as experienced by conventional CPLs[4]. Even if the wick survived the high superheat start-up, preventing high vapor velocities venting out of the liquid-blocked channel would be a challenge. A second potential problem with the gravity-assist LHP was that, at low power input, the total system pressure drop was not sufficient to overcome the hydrostatic pressure head. As a result, liquid would partially fill the evaporator vapor channel causing a temperature gradient that could exceed 3°C. Finally, there was concern in regard to the fluid distribution arrangement for the returning fluid. As a result of the tolerance variations inherent in manufacturing, it was postulated that the returning fluid would preferentially feed several evaporators, leaving one or more to dryout.

WORKING FLUID SELECTION

Working fluids found in capillary pumped systems must provide high heat transfer properties and high surface tension capability. For the operating range dictated for this application, ammonia, acetone, methanol, and water were all considered. Ammonia has a high vapor pressure that could not be contained within the flat plate geometry without excessive wall thickness. Acetone and methanol would be acceptable, except that toxicity and flammability considerations made these fluids less desirable. Water was found to satisfy the design constraints, particularly in light of the fact that the unit would only operate in a gravity environment, as will be further explained later. However, below 80°C, vapor density ~~is~~ so low that the vapor flow is susceptible to flow choking (sonic limit).

MATERIAL SELECTION

With the decision to use water, the selection of a material with acceptable corrosion resistance and high conductivity led to copper as the most acceptable material for all components of the system. Copper/water systems have been found to be the most compatible from a non-condensable gas generation standpoint for conventional heat pipes[5]. Copper also provides a highly machinable material for ease of fabrication.

EVAPORATOR DESIGN

Design of the evaporator section had to ensure:

- 90 percent of the area was available for heat removal
- Adequate distribution of returning liquid
- Adequate vapor space to ensure sonic limits were not approached
- Sufficient capillary pressure to pump flow through the system
- A fairly rugged design capable of meeting the flatness requirements

The basic design of the evaporator consisted of two copper plates with a sintered copper wick sintered in each plate. The vapor space was formed in the middle once the plates are joined. Structural supports were placed within the vapor space to prevent collapse of the shell under operating conditions, less than ~~1 atm~~ to allow for operating ~~temperatures~~ below 100°C. An irrigation tube placed within the evaporator vapor space and along the top edge provided fluid distribution. Small holes were evenly spaced along the length. Fluid returning from the condenser would drip out of these holes and feed the surface of the wick. Details of the evaporator design are shown in Figure 5.

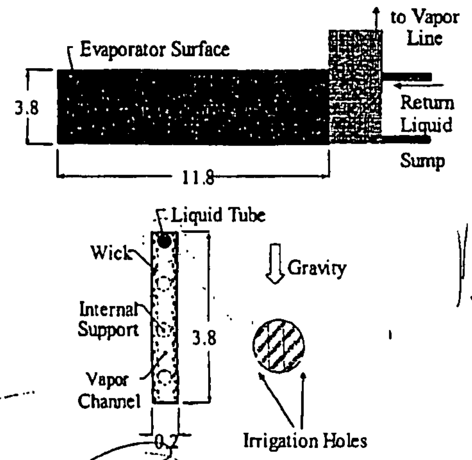

Figure 5 – Thin Flat-Plate Evaporator Design

With the total thickness of the section limited to 1.3 cm, the shell, wick, and vapor space thicknesses were highly constrained. A shell thickness of 0.24 cm for each face plate was determined to be required to provide sufficient stiffness to meet flatness requirements. As a rule of thumb, it was desirable to maintain the local Mach number of the vapor to less than 0.2. This was considered to be necessary to provided sufficient margin to the sonic limit in light of the numerous internal supports. Thus, a vapor space of 0.6 cm was determined to be necessary. This left a wick thickness of 0.08 cm on each faceplate.

The thin wick applied to the faceplate ensured even distribution of fluid across the heat transfer surface to ensure isothermal conditions. Copper powder was sintered on both sides of the evaporator. A small pore size between 20 and 25 microns was necessary to provide a sufficient capillary pumping head to maintain system flow. High porosity was desired, in excess of >40 percent to minimize the internal flow resistance. The sintering parameters chosen where intended to provide the desired pore size and porosity. These parameters were not sufficient to ensure sufficient capillary action at full power operation. Operation in a gravity environment, however, provides sufficient pressure to augment the capillary pumping head at high power operations.

MULTI-EVAPORATOR CONSIDERATIONS

An important aspect of the experiment was whether it is practical to network several evaporators together to utilize a common, water-cooled condenser. To accomplish this goal, the vapor and liquid return lines were manifolded together, as shown in Figure 6. The vertical location of the liquid manifold was chosen such that the individual lines would remain above the end of the evaporator when the unit was inclined 45°. Additionally, the evaporator sumps were manifolded together to allow liquid excess to equalize between evaporators. The use of these manifolds was judged to provide some assurance that fluid would be available to each evaporator regardless of unique evaporator operating characteristics. Placement of the vapor line manifold was not considered critical.

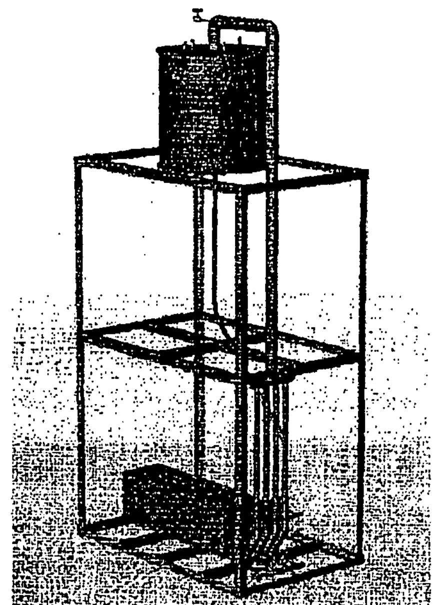
Figure 6 – Layout of Four-Evaporator LHP

Previous efforts at networking multiple evaporators and condensers have been successful[7, 8]. However, these systems include a reservoir for each evaporator that would have minimized fluid distribution concerns. Addition of a reservoir was not desirable to maintain a relatively simple design.

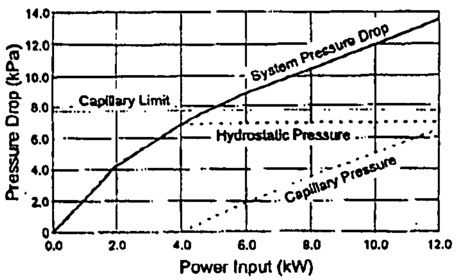

Figure 7
Pressure Drop Analysis for Single Evaporator

ANALYSIS

STEADY-STATE PRESSURE DROP ANALYSIS

A pressure drop model originally developed to calculate the frictional flow losses in high pressure LHPs was modified for low-pressure fluids[1, 6]. Dimensions of the components are shown in Table 2 and predictions of the LHP are presented as functions of power input in Figure 7. As expected, the total system pressure drop at low power (<16 kW or 4 kW per evaporator) was not high enough to prevent liquid from partially flooding the evaporator. Below 16 kW the fluid flow was supported entirely by a hydrostatic pressure head (capillary action did not exist since liquid flooded the wick). Above a power input of about 19.2 kW, capillary action alone was not adequate to overcome the system pressure drop. The analysis indicated that the LHP required an additional gravitational head equivalent to a minimum vertical distance of 34 inches between the condenser and the evaporator (condenser above evaporator) for the LHP to work at maximum power of 48 kW.

Table 2: Components of Multiple-Evaporator LHP

Component	Dimension
Vapor Line	
Outer Diameter	5.1 cm
Wall Thickness	0.08 cm
Length	305 cm
Liquid Line	
Outer Diameter	1.91 cm
Wall Thickness	0.08 cm
Length	305 cm
Condenser	
Outer Diameter	50.8 cm
Wall Thickness	0.32 cm
Length	50.2 cm
Evaporator	
Length	76.2 cm
Width	24.4 cm
Overall Thickness	1.3 cm
Vapor Channel Width	15.2 cm
Vapor Channel Depth	0.6 cm
Liquid Tube Length	96.5 cm
Liquid Tube Orifice	0.08 cm
Wick Thickness	0.16 cm
Wick Permeability	~5.0E-12 m^2
Wick Pore Radius	~20 microns

TRANSIENT ANALYSIS

A thermal LHP model was developed to simulate the start-up dynamics and transient behavior with respect to changes in the operational conditions. SINDA85, a FORTRAN based thermal/hydraulic analyzer, was utilized. LHP temperatures for a single evaporator are shown in Figure 8 during a start-up to full power in 4 kW per evaporator increments. As shown, the LHP saturation temperature increased very quickly until it reached 51°C. Soon after this, the LHP evaporator temperatures dropped back to 44°C, before approaching a steady-state temperature of 47°C.

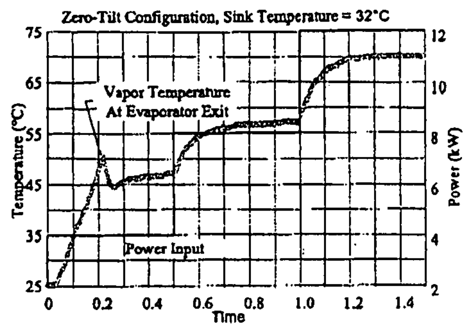

Figure 8 - Predicted Response to Start-Up Transient

This start-up phenomenon is described as follows:

- Prior to start-up, the vapor channels of the LHP evaporator were partially filled with liquid
- As power was applied to the evaporator, the combination of an initially low vapor density (temperature ~20°C) and a liquid-blocked vapor channel created a vapor-choked condition (sonic limit) that did not allow the entire heat input to be

transported to the condenser. As a result, the evaporators heated up rapidly with the excess heat that was not transported to the condenser.

- When the evaporators reached a temperature at which the vapor density was high enough and vapor choking no longer existed, vapor rushed out of the evaporator and flowed quickly to the condenser creating a sharp decrease in the saturation temperature.
- Evaporator temperatures gradually increased until a steady-state condition of 47°C was reached.
- Following the start-up process, the power input increased to 32 kW and 48 kW. The response was very similar to any other LHP system.

MOCK-UP FABRICATION

A single-evaporator gravity-assist LHP test loop was built and performance tested to serve as a pathfinder for the proposed multiple-evaporator system[1]. The loop was tested in three tilt configurations: horizontal evaporator, evaporator tilted +45° along the long axis, and evaporator tilted –45°. In each tilt configuration, the test loop was subjected to a series of LHP tests: start-up, power cycling, high power, low power, and variation of sink conditions. It performed extremely well in all phases of the test program.

DESCRIPTION OF TEST PROGRAM

Testing of the multiple-evaporator gravity-assist LHP test loop consisted of normal and quick start-ups, low and high power runs, and uneven heating runs. The main objectives were to (i) determine the system operational limits in different tilt conditions and, more importantly, (ii) characterize the thermodynamics and fluid dynamics interaction among the evaporators.

Lessons learned during testing of the mockup unit facilitated the test preparation and fluid charging. Initial calculations indicated that a minimum charge of 8 gallons of water was required to prevent vapor from leaving the condenser in ±45° tilts. The unit was actually charged with 12 gallons of water before the testing began.

Two aluminum plates containing cartridge heaters were clamped to both sides of each evaporator to simulate the system heat load. Each heater plate could generate a power input in excess of 6 kW. Cooling water was circulated in the heat exchanger coil to provide heat rejection for the test loop. Twenty-four Type T thermocouples were used to monitor the loop temperatures during the test. Most of the thermocouples were attached to various locations on the evaporator so that the surface temperature gradient could be easily determined. The thermocouple readings were recorded by a PC-based data acquisition system every 5 seconds.

TEST RESULTS

Horizontal Tests -

In this test configuration, the unit was subjected to various tests including start-ups, high power, long-duration low power, power cycling, and uneven heating among the evaporators. The unit worked very well in all test conditions. It should be noted that the heat transport of 48 kW could not be demonstrated for more than 15 minutes due to the electrical circuit breaker limits of the test facility. Further testing will be performed at full power for extended periods at an alternate test facility. All figures presented in the following sections provide data for a single evaporator.

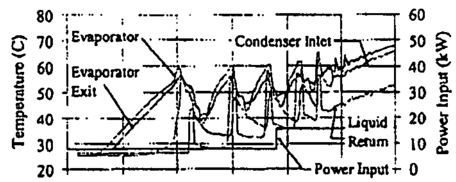

Figure 9 – Start Up Transient

Start-ups - The temperature response of a typical start-up is shown in Figure 9. In this test, a power input of 2 kW was applied evenly to all four evaporators (8 kW total) at 11:30. Temperatures of the evaporator and the manifold increased quickly and at the same rate indicating that evaporation had occurred inside the evaporators. However, vapor temperature at the condenser inlet remained unchanged (~26.6°C) for more than 10 minutes. During this time, vapor did not flow in the vapor line to the condenser. A combination of initially liquid-blocked vapor channels and low vapor density at low temperatures created a difficult condition for vapor to vent out of the evaporators. Not until 11:40 did the evaporator temperature reach 60°C and, more importantly, vapor build up sufficient pressure for the vapor to spurt out lowering the evaporator temperatures and sharply increasing the condenser inlet temperature at 11:41. Finally, vapor flowed (with a high mass flow rate) in the vapor line to the condenser. But the vapor transport was short-lived because, at 11:42, the evaporator temperatures went up again while the condenser inlet temperature decreased and then remained constant around 28.5°C. This indicated that vapor stopped flowing and the

temperature and pressure began to build-up for the next spurt. The cycle repeated itself at 11:45 and again at 11:48. It was speculated that if the operating conditions (i.e. power input at 2 kW/evaporator and coolant flow rate at 1 gallon/minute) had remained unchanged, it would have been possible that the periodic spurts of vapor persisted (pulsating flow) and a steady state could never be reached.

As shown in Figure 9, at 11:49, the power input increased to 4 kW/evaporator (16kW total). Four noticeable cycles of vapor spurting continued. However, the intensity (i.e. temperature fluctuations) diminished successively as the evaporator temperatures increased. The vapor spurting finally disappeared when the evaporator temperatures increased above 65°C. Beyond this temperature, vapor density of water was sufficiently high to overcome the vapor "choke" condition in the evaporator (as discussed in the Transient Analysis section). This allowed vapor (or waste heat) from the evaporators to flow continuously (without hiccups) to the condenser for heat rejection.

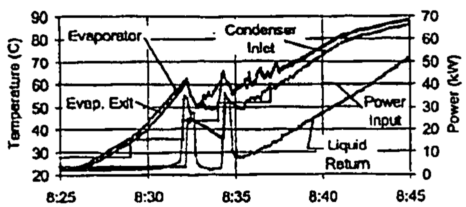

Figure 10 - Quick Startup

Quick Start-ups - In the quick start-up procedure, 2 kW/evaporator of heater power was applied to all four evaporators (8 kW total). Power was then stepped up by 2 kW increments every two minutes until it reached 10 kW/evaporator (40 kW total). As presented by the test data in Figure 10, the same start-up phenomena (described above) were observed in this test regardless of the power input. At first, evaporation occurred inside the evaporators but vapor did not flow to the condenser. The evaporator temperatures exceeded 60°C at 8:32 to induce the first spurt of vapor out of the evaporators. Vapor spurts repeated again several times after that but disappeared when the evaporator temperatures increased beyond 70°C.

High Power - Figure 11 shows the temperatures during the high power test. To start the test, 2 kW/evaporator was applied to all four evaporators (8 kW total). Power was then increased in 2 kW increments to 12 kW/evaporator (48 kW total). Even though analytical model predictions indicated that water LHPs could function properly with vapor temperature maintained at least above 62°C, the vapor temperature in this test was kept between 70°C and 80°C by throttling mass flow rate of the coolant flow. Once the periodic vapor spurting in the loop start-up stopped, the unit performed very well in terms of heat transport capability (>48 kW) and temperature gradient (<3°C) over the evaporator area.

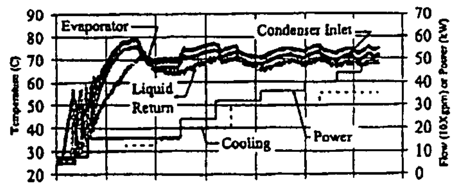

Figure 11 - High Power Test

Long-Duration Low Power - Following a 2 kW/evaporator start-up, power was reduced to 0.5 kW/evaporator (2 kW total) and maintained at this level for at least 7 hours. Test results are shown in Figure 12. During low power operation, the coolant mass flow rate was varied to investigate effects of the vapor temperature on operation. As expected, at a power level of 0.5 kW/evaporator, liquid partially blocked the evaporator vapor channels. Fluid dynamics of shear flow between the vapor flow (venting out of the evaporator) and the liquid puddle created an unsteady evaporative heat transfer inducing a loop temperature oscillation in the process. Amplitudes of the oscillation grew larger when the loop temperature was lowered (60-70°C) by an increase of the coolant mass flow rate.

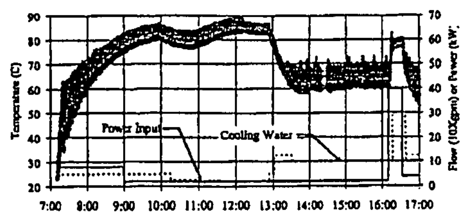

Figure 12 - Low Power Long Duration Test

Uneven Heating - Test results of the uneven heating are shown in Figure 13. In this test, the power input started at 10 kW/evaporator (40 kW total). Evaporator A power was then reduced by 2 kW incrementally until the power level reached 2 kW. Likewise the same power reduction steps were repeated for the remaining evaporators until power inputs on all four evaporators reached 2 kW/evaporator (8 kW total). After that, each

evaporator power was increased to 12 kW/evaporator (48 kW total) in 2 kW increments.

The unit performed well in this test meeting the temperature gradient requirement (<3°C) for power level of individual evaporators above 4 kW. When an evaporator power dropped below 4 kW, the temperatures fluctuated for the same reasons described in the preceding sections.

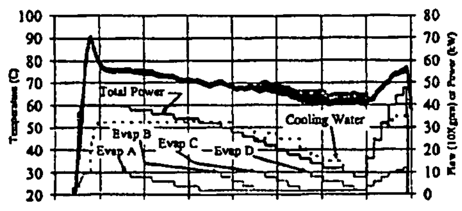

Figure 13 - Uneven Heating Test

Long-Term High Power - In this test, a power level of 10 kW/evaporator was maintained on each evaporator (40 kW total) for a minimum of 6 hours. During the test, loop temperature was successfully controlled between $67^{o}C$ and $85^{o}C$ by varying the coolant mass flow rate. Further testing will be performed at full capacity and for longer periods of time.

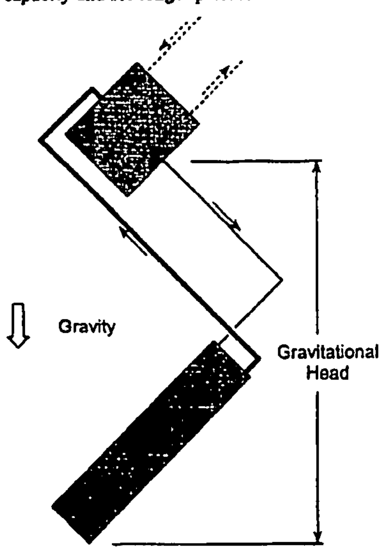

Figure 14 – +45°

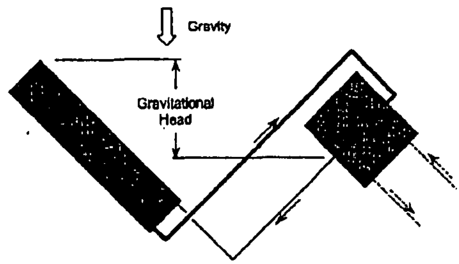

Figure 15 – -45°

Tilt Tests

The unit was tilted +45° and –45° along the long axis of the evaporators, as shown in Figures 14 and 15.

Start-ups - Start-ups were carried out with 2 kW/evaporator applied to all four evaporators (8 kW total). Figure 16 shows the temperature response during the start-up process in the +45° tilt configurations. The same periodic vapor spurts (out of the evaporators) in early stages of the loop start-up were also observed in this test even though it took longer for the spurts to disappear.

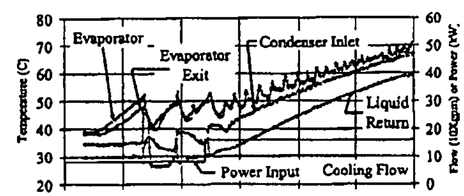

Figure 16 - Inclined (+45°) Startup

Loop Operation - In this test, power input to each evaporator was stepped by to 11 kW/evaporator (44 kW total) following a 2 kW/evaporator start-up. Temperatures for Evaporator C are shown in Figure 17 for this test. At 16:26 when the power level was 11 kW/evaporator, one of the temperatures on Evaporator C ran away indicating a partial deprime on that evaporator. All other evaporators still functioned normally when this happened. At 16:28, power was reduced to 8 kW/evaporator (32 kW total) in an effort to reprime the Evaporator C wick. The Evaporator C temperature that ran away in the previous power level immediately dropped back to the same level as the rest of Evaporator C temperatures suggesting a reprime of

its wick. To make sure the Evaporator C wick completely recovered, power was further reduced to 4 kW/evaporator (16 kW total) at 16:31 and maintained at this level for 15 minutes. The coolant mass flow rate was adjusted to keep the loop vapor temperature above 70°C. At 16:48, power input was brought back to 11 kW/evaporator (44 kW total). After 30 minutes, none of the evaporators showed any sign of deprime. At 17:15, the total system power was increased to ~48 kW and maintained at this level for 15 minutes. All temperatures were steady and within specification during this period.

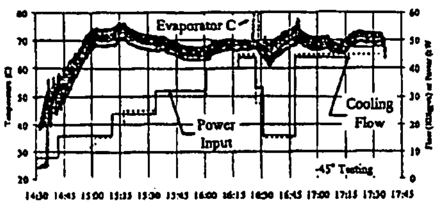

Figure 17 - Evaporator C Response

-45° Tilt Tests

The unit was also tested with the evaporators tilted -45° along the long axis as.

Start-ups - Again start-ups were carried out with 2 kW/evaporator applied to all four evaporators (8 kW total). Figure 18 presents the temperature response during the start-up process in the -45° tilt test. The same periodic vapor spurts (out of the evaporators) were also observed in this test especially at low power level (i.e. 2 kW/evaporator). The vapor spurts seemed to be more intense than those in either the horizontal or +45° tilt tests. Note that liquid might completely block the evaporator exit requiring more pressure buildup in the evaporator vapor channel for vapor venting to happen.

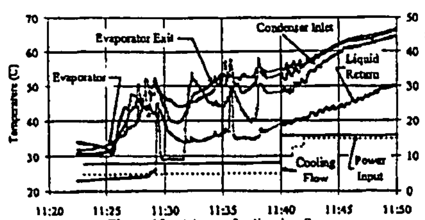

Figure 18 - Adverse Inclination Startup

Loop Operation - Figure 19 presents the evaporator temperatures in this -45° tilt test. Following a 2 kW/evaporator start-up test, power input was increased to 4 kW/evaporator (16 kW total) at 11:40 and again to 6 kW/evaporator (24 kW total) at 12:06. Again the coolant mass flow rate was regulated to control the vapor temperature. The loop functioned normally (without noticeable anomalies) at power level of 4 kW/evaporator. The loop also worked fine for the first 15 minutes after the power input was increased to 6 kW/evaporator. Note that during this time period, the loop vapor temperature was maintained at 70°C with a coolant mass flow rate of 2.5 gallons per minute. Only when the coolant mass flow rate was raised to 3 gallons per minute and the loop temperature dropped to 65°C, did Evaporator C showed signs of a wick dryout. The remaining evaporators continued to operate without a problem. The -45° tilt test series, however, ended without any attempt to reprime Evaporator C wick. Further testing is required to verify operation at this condition.

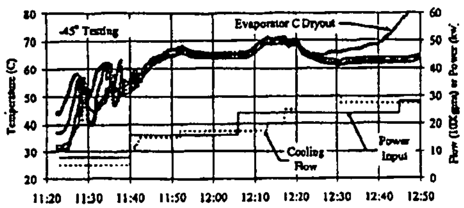

Figure 19 - Evaporator C Dryout

CONCLUDING REMARKS

A gravity-assist LHP using water as the working fluid successfully demonstrated that the capillary pumped heat transport technology was capable of providing large-scale waste heat removal. The unit met all expectations at horizontal test conditions. Further testing while the unit is adversely inclined is required.

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